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BENDING FATIGUE FAILURES OF COMPRESSOR VALVE REEDS

S. Papastergiou, PhD

ABSTRACT

Bending fatigue failures of cantilever type suction valve reeds occurred under operating conditions in a semi-hermetic refrigerant compressor during field trials. The finite element method (FEM) was applied to predict both the static and dynamic displacements and the resultant stresses of the reeds. The dynamic stresses in the location of the fracture exceeded the fatigue limit of the reed steel. The procedure could assist in the modification of a reed design so that failures due to bending fatigue were avoided.

INTRODUCTION

The self-acting valves in reciprocating compressors are perhaps the least reliable components, especially at the higher compressor speeds now being introduced in larger compressors.

A finite element model (1,2) was developed to evaluate the stresses in cantilever type suction valve reeds due to bending. Impact stresses were not considered. Ab initio displacements and stresses under static conditions were calculated. Some useful conclusions were drawn from this initial part of the study despite the neglect of dynamic effects.

STATIS ANALYSIS

Figure 1 shows the finite element model and static stress distribution along the centre line of a cantilever type suction valve reed. Since the stress field predicted by the FEM is two-dimensional, equivalent stresses (\(\sigma_{eq}\)) at the port centre and the root of the reed are also indicated in Figure 1. When a relatively fine grid with 93 degrees of freedom (d.o.f.) is used, the predicted stresses are higher than when a cruder grid with only 39 d.o.f. is employed.

At the root of the reed the static equivalent stresses were greater than the bending fatigue limit of the reed steel, although much less than the ultimate tensile strength of the steel. The bending fatigue limit was approximately 720 N/mm² for the stainless steel (SANDVIK 7C27Mo2) employed. The predicted equivalent static stresses at the root of the reed were 805 N/mm² and 942 N/mm² using the crude and fine grids respectively. At the centre of the port, the predicted static equivalent stresses were less than the bending fatigue limit of the reed steel, when using either grid size. Hence the static analysis suggested that the reed might fail at its root; During field trials, under normal compressor operating conditions, failures did occur near the root.

However, predictions of reed failure due to bending fatigue cannot always be made by such a comparatively simple static analysis. For example, breakages had also occurred under operating conditions when the pressure difference across the reed was considerably less than the value under which the analysis in Figure 1 (2) was made. In these cases the predicted static bending stresses were less than the bending fatigue limit of the reed steel. These failures might be predicted by a dynamic analysis, since dynamic stresses are always much greater than the equivalent static stresses (2). Moreover, a dynamic analysis of the motion can predict the detailed reed displacement which in turn may affect the compressor performance by influencing valve flow areas. Hence a dynamic analysis is justified despite the higher demands on computer resources (2).

DYNAMIC ANALYSIS

Figure 2 shows the predicted maximum dynamic stress distribution along the centre line of the cantilever reed, when two grid sizes were employed. Using either grid size, peak stresses at root and port centre were greater than the fatigue limit of the reed steel; the static analysis had predicted stresses greater than this limit only at the root of the reed. The values of maximum equivalent dynamic stress were such that the reed was likely to fail first at the root: If a stress raiser occurred near the centre of the port, the reed might fail first near that location. During field trials, these reeds did fail near the root, but occasionally they also failed near the port areas (3).

When the thickness of the reed was decreased from 0.7mm to 0.45mm the simulation model (4) showed
that the power losses due to throttling across the valve were reduced, the permitted lift at the reed tips remaining unaltered. The dynamic stresses which occurred at the centre of the port now increased and exceeded the stresses at the root of the reed suggesting that the thinner reed might fail first near the centre of the port. However in both locations the dynamic stresses were again greater than the fatigue limit of the reed steel. The values and locations of dynamic stresses near the reed roots were such that the life of the thicker reed was likely to be about 30 minutes \((8 \times 10^4\) cycles) from S-N curves for the reed steel. During field trials, the thicker reeds did fail near their root after approximately two days \((6 \times 10^6\) cycles) of operation, while the thinner reeds also failed quickly but with the fracture near to the valve port (Figure 3). Ideal clamping conditions had been assumed in this finite element model but it was considered that the rubber pads at the clamping areas allowed some degree of rotation, (imperfect clamping, Refs. 1, 5) Thus the actual stresses at the root of the reeds would be reduced so accounting for the observed longer life of the thicker reed. Allowance for this imperfection of clamping has been made in the finite element model (1,5) but was not included in the present study, due to the additional demands on computer resources. While the imperfection of clamping reduced the stresses at the root of the reed it would be expected to increase the stresses at the port centre, possibly also explaining why the thicker reeds occasionally failed first near the location of the port.

SUCTION REED STRESSES DURING THE DISCHARGE PROCESS

The maximum static stresses were evaluated in the area of the port of the 0.7mm thick suction reed when being "punched" into the port during the discharge phase of the compressor cycle (Figure 4). The area of the reed over the port was treated as a circular plate simply supported at its edges and uniformly loaded. In practice the conditions at the boundaries are somewhere between clamped and simply supported, but closer to the latter. For a constant condensing pressure of 23 bar and an evaporating pressure of 6 bar, the maximum stress at the centre of the port was about 250 N/mm\(^2\). This static value would rise to 380 N/mm\(^2\) due to a peak cylinder pressure of 31 bar which occurred during the early part of the discharge phase. These values of maximum static bending stress when "punching" the suction reed into the port during the discharge phase were much less than the maximum dynamic bending stresses in the reed during the suction phase of the same compressor cycle.

The corresponding maximum static stresses at the centre of the port for the thinner (0.45mm thick) suction reed during discharge were 605 and 920 N/mm\(^2\). These static values although high, were less than the maximum dynamic bending stresses in the thicker reed at the centre of the port during suction phase. Hence the static stresses due to punching of the thinner reed into the port during discharge were also less than those due to dynamic bending during suction of the thinner reed. Thus even if the thinner suction reed is expected to fail first near the centre of the port due to bending fatigue during suction, the stresses (920 N/mm\(^2\)) predicted at the same location during discharge are also high and exceed the fatigue limit.

The finite element model developed (1) could also be applied to predict the dynamic behaviour of suction reeds when they are pressed against the valve port during compression, discharge and re-expansion. The demands on computer resources would be great due to the large number of steps needed in the integration procedure, and the much finer grids required in the area of the valve port (1). Thus the present study of the "punching" of the suction reed into the port was limited to a static approach.

Transient temperature effects on valve reeds during compressor operation may superimpose thermal stresses. Calculations showed that such additional stresses due to thermal cycling were small compared to the bending stresses, due to low Biot No. (3). Brown and Lough (6) applied laser holography to study the distortions due to temperature effects on stainless and mild steel disc valves.

By employing such a finite element model, the design of a suction reed which fails due to bending fatigue during suction may be modified to reduce the dynamic stresses being generated. Subsequently the effect of such modifications to the valve design on the performance of the compressor can be estimated by a suitable simulation model (4,7). In the cases studied, the stresses were avoided by decreasing the reed length by approximately 20% and the port diameter by 6%. The reed thickness was kept at 0.7mm but the shape was altered slightly (2). The increased stiffness of the reed resulted in maximum dynamic stresses less than the fatigue limit of the reed steel. The compressor performance did not deteriorate significantly. Thus the final design of the reeds was such that failures were avoided. (2)

CONCLUSIONS

Computer programs which applied the finite element method predicted the static and dynamic stresses of a cantilever type of suction valve reed. Maximum dynamic stresses at the centre of the port were about 50% greater than the equivalent static values with the same pressure difference across the reed at the same port. Hence dynamic analysis is justified despite the higher demands on computer resources.

The predicted maximum dynamic stresses in the situation examined were greater than the fatigue limit of the reed steel. Failures were predicted to occur near the root of the thicker (0.7mm) reed and near the port area for the thinner (0.45mm) reed. Field trials substantiated these predictions. A consequence of the study was that the finite element model provided an aid to the design of modified reeds which avoided the failures that had been occurring.
APPENDIX

FRACTOGRAPHIC ANALYSIS OF FRACTURED REEDS

A fractographic investigation of reeds which had failed during field trials was made using a Philips PSEM 500 scanning electron microscope. The hardness of the martensitic stainless steel was 572 H.V. measured by a Leitz microhardness tester (applied force 1N). Plates 1 and 2 show parts of the surface of the fatigue fracture. A smearing effect can be discerned over some of the surface: This may have occurred after the failure. The contours of the fracture striations indicate that the failure initiated near the edge of the reed. This may be attributed to the stress enhancement which occurs at the edges of the reed as a result of transverse bending. In the reeds analysis (Figure 1) the maximum deflection at the edges of the thick reed due to such transverse bending is approximately 0.07mm, causing an increase in local stresses of about 5 - 10% (8,9). This effect would be accounted for in the finite element model only if computer resources permitted the use of very fine grids.

In Plate 3 two types of fracture striation may be seen, ductile in the centre of the reed (see also Plate 4) and brittle close to the surface of the reed which is subject to work hardening.

The exact location of initiation of the fracture could not be specified. Laub (10) claims that bending failures are likely to initiate just below the surface of a reed (0.025 to 0.05mm) due to the presence of residual stresses.

Gouges, pits, surface imperfections and non-metallic inclusions together with surface wear and fretting may accelerate failures (8,10). The steel used in high quality so these faults are not likely to be significant stress raisers, particularly in the good lubrication and non-oxidising environment in which most refrigerant compressor reeds operate. High dynamic stresses due to flexing or to punching into a port are the most probably causes of bending fatigue failure. Such failures are most likely to be initiated at the edges of the reed.

REFERENCES


1.0 X/t

Fig. 1. Static stress distribution along $L$ of a suction valve reed.

Fig. 2. Maximum dynamic stress distribution.

Fig. 3. Locations of maximum stress and fracture.

Fig. 4. Suction reed is pressed into suction port during discharge phase of compressor cycle.

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**Fig. 1**

Static stress distribution along $L$ of a suction valve reed.

**Fig. 2**

Maximum dynamic stress distribution.

**Fig. 3**

Locations of maximum stress and fracture.

**Fig. 4**

Suction reed is pressed into suction port during discharge phase of compressor cycle.

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**Table 1**

<table>
<thead>
<tr>
<th>Location</th>
<th>Maximum Stress</th>
<th>Location of Fracture</th>
<th>Maximum Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location of Fracture</td>
<td>0.7 mm Thick</td>
<td>Location of Fracture</td>
<td>0.45 mm Thick</td>
</tr>
</tbody>
</table>

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**Equations**

For $N$ degrees of freedom (dof):

- Port Centre
  - $\sigma_1 = 762$
  - $\sigma_2 = 129$
  - $\sigma_{eq} = 723$

- Root of the Reed
  - $\sigma_1 = -1100$
  - $\sigma_2 = -100$
  - $\sigma_{eq} = -1130$

- Port Centre
  - $\sigma_1 = 984$
  - $\sigma_2 = -36$
  - $\sigma_{eq} = 995$

- Port Centre
  - $\sigma_1 = 1280$
  - $\sigma_2 = 65$
  - $\sigma_{eq} = -1296$
Plate 1 Reed fracture initiated in the vicinity of the reed edge (X20).

Plate 2 Reed fracture in the vicinity of the reed edge (X 80).

Plate 3 Brittle and ductile type of fracture (X 160).

Plate 4 Ductile type of fracture (X 160).